

**OPTIMIZATION OF ROTOR BLADES FOR COMBINED STRUCTURAL,
PERFORMANCE, AND AEROELASTIC CHARACTERISTICS**

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ABSTRACT

This paper outlines the strategies whereby helicopter rotor blades can be optimized for combined structural, inertial, dynamic, aeroelastic, and aerodynamic performance characteristics. There are three key ingredients in the successful execution of such an interdisciplinary optimization. The first is the definition of a satisfactory performance index that combines all aspects of the problem without too many constraints. The second element is the judicious choice of computationally efficient analysis tools for the various quantitative components in both the cost functional and constraints. The third element is an effective strategy for combining the various disciplines either in parallel or sequential optimizations.

INTRODUCTION

This paper describes ongoing work in the optimization of helicopter main rotor blades. The helicopter is intrinsically interdisciplinary due to the close coupling between aerodynamics, dynamics, and the blade structural details. As a result, the optimization of a helicopter rotor involves important design considerations from many diverse engineering disciplines. In our present work, we attempt to combine several of these important effects in a unified manner. First, the blade must be designed to have natural frequencies that are removed from integer multiples of the rotor speed. This is necessary in order to ensure good dynamic characteristics. Second, the blade must be as light as possible but yet have sufficient inertia to allow autorotational landings. Third, the structure must be designed to ensure that blade stresses can be safely carried by the cross section through an adequate number of loading cycles. Fourth, this cross-sectional structure must fit within the aerodynamic envelope of the blade and be manufacturable. Fifth, that aerodynamic envelope must yield satisfactory rotor performance in hover and forward flight. Sixth, the combined structural, inertial, and aerodynamic characteristics of the blade must be aeroelastically stable with low vibrations. In the present work, we concentrate on the best methods of formulation for the above considerations such that the blade can be optimized effectively.

There has been a good deal of good work in rotor optimization through the years. Although we do not have space to do a complete survey, certain important contributions should be mentioned. In ref. 1, it is noted that efficient placement of lumped masses within the blade can lower helicopter stresses. References 2 and 3 attempt optimum placement of these masses but note that the minimum vibration solution often results either in high blade stresses or in bending-torsion flutter due to the natural migration of the torsional frequency to an integer multiple of rotor speed during the optimization process. Bielawa, ref. 4, performs a "man-in-the-loop" optimization in which aeroelastic stability is an integral part of the problem. He notes that a major hindrance to completely automated optimization is the complexity of constraints that define a realistic blade design.

In more recent work, Taylor (ref. 5) shows that optimization to low vibrations can result from efficient tailoring of mode shapes as well as from frequency placement. Reference 6 provides a combined aeroelastic and vibration optimization with complete flap-lag-torsion equations. The work shows that such an optimization is feasible, but that frequency constraints must still be applied in order to prevent migration of some modes to undesirable values. Reference 7 provides an optimum placement of dynamic frequencies based on initial designs of in-service rotor blades. Design parameters are taken to be the internal structural thicknesses of box beams and lumped masses (rather than the generic EI's, etc., used in previous work). Results show that all frequencies can be effectively placed with the use of realistic

structural changes that can fit within the aerodynamic envelope. Reference 8 presents work on the optimization of rotor blades in order to have good aerodynamic performance, a consideration that is lacking in the earlier structural optimizations.

In the most recent work, refs. 9 and 10 re-examine the optimization problems of refs. 5 and 7 but with emphasis on optimization strategies and use of limited design spaces. Reference 11 provides some of the most involved aeroelastic optimization to date. This research shows the importance of obtaining analytic modal gradients in order to make the optimization procedure efficient. Finally, ref. 12 offers the first experimental verification that numerical optimization can truly result in rotor blades with lower vibrational characteristics.

In this paper, we look at the theoretical and practical problems associated with the addition of stress constraints and aerodynamic performance goals to the traditional structural and aeroelastic optimizations listed above. In traditional design methodologies in the industry, the choice of the aerodynamic blade shape (chord, thickness, twist, etc.) is the first step in the design process. This geometry is chosen based on aerodynamic performance considerations. Next, a structural design is performed in order to find an adequate structure (i.e., one that can withstand the blade fatigue loads) that can fit within the aerodynamic surfaces. Third, aeroelastic and vibrational analyses are performed to see if the blade needs to be tuned further in order to eliminate either instabilities or harmonic resonances (the latter of which would impact the stress calculations). It may very well be that a more "optimum" design could be obtained if these various individual optimizations were done in parallel rather than in series. For example, it may be that a slight compromise in blade performance (in order to accommodate additional structure) might lower vibrations to the point that a heavy vibration absorber could be eliminated, thus mitigating the performance loss. Therefore, it is important to determine how such a unified optimization might be performed.

THEORETICAL BACKGROUND

Structure

The first step in the optimization research described herein is to replace the true blade with a realistic, box-beam model that has dynamic characteristics similar to that of the true beam as well as a realistic stress distribution. Figure 1 depicts the schematic model used here. The blade chord and airfoil thickness are assumed to come from a performance analysis which could be running in parallel or in series with the structural analysis. This, then, defines a geometric area within which the box beam may be placed. The primitive design variables for the box beam are its width (b), its flange thickness (t), and its web thicknesses (s_1 and s_2).

The box beam is assumed to carry all blade tension and all vertical bending. However, the secondary cell formed by the trailing-edge skin is assumed to contribute to both torsional stiffness and inplane stiffness through a skin thickness (p). Two additional primitive design parameters (a and d) allow for additional freedom in weight distribution. The parameter (a) defines the size of the tip weight, and the parameter (d) defines the width of a lumped mass internal to the box beam. These seven design parameters (along with given material properties) define the blade mass and structural properties. Naturally, they are constrained such that the pieces must fit within each other (e.g., $d < b - s_1 - s_2$).

Now, these seven primitive design variables translate into eight overall structural properties. These are the two bending stiffnesses (EI_f and EI_c), the torsional

Elements of Analysis

Once the structure is defined, the next step is to set up the analysis tools required in the optimization process. These are listed below:

1. Weight and inertias
2. Natural frequencies and modes
3. Performance and handling qualities
4. Vibrations and loads
5. Blade stresses and fatigue life
6. Aeroelastic stability

In the first category, a calculation of blade weight is necessary because blade mass not only adds its own weight to the helicopter but also results in additional weight in the control system, bearings, etc. Thus, every pound of blade weight could result in 3 pounds of structural weight, which implies less payload, more fuel, or shorter range. The mass moment of inertia is important because it must be large enough to support autorotation. For example, some companies recommend that the kinetic energy in the blade divided by hover power should be at least 2 sec. Also, the chordwise mass balance is important to vibration and to bending-torsion flutter. All three of these aspects are included in the present work.

In the second item, we find that the natural frequencies and mode shapes can also enter the optimization process. Past optimization studies have found it necessary and useful to keep blade frequencies within prescribed bounds. Figure 2 from ref. 7 shows vertical shears as a function of the second natural frequency of a teetering rotor (symmetric mode). One can see the strong coupling between frequency

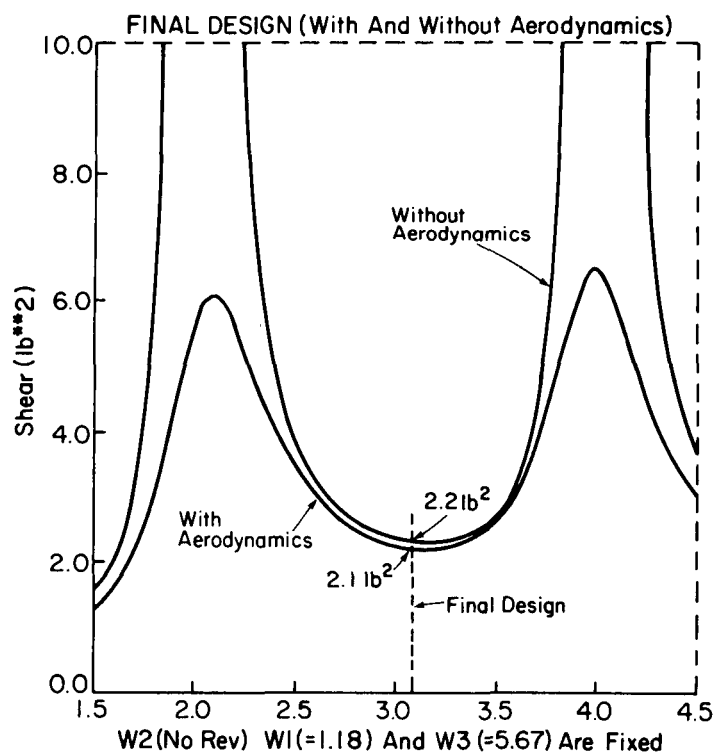


Figure 2. Effect of Frequency on Shear Stress

placement and vibrations both with and without the aerodynamic damping. This is one reason that frequency placement has always been a high priority of blade designers. Furthermore, as pointed out in refs. 5 and 12, the mode shape, itself, also has a strong influence on vibration because it affects the generalized force of each mode. In addition, poor placement of lower-frequency modes can adversely affect handling qualities; while proper placement of higher-frequency modes can improve performance. In this work, we use a subspace-iteration method to find the modes and frequencies of the structure. In the initial phase of optimization, frequency placement is part of the objective function such that frequencies begin to move within some prescribed window. Once any frequency falls within this window, however, its placement is then switched to a constraint. This process allows us to overcome problems associated with an infeasible initial guess. Table 1 summarizes the eigen analysis.

The third item deals with the performance of the aircraft. This area is very sensitive to the mission of a vehicle, and the "optimum" performance depends on the mission profile. In order to gain insight into multidisciplinary optimization without being mired in undue computations, we have chosen to optimize for the best hover performance out of ground effect. Admittedly, this is a very simple beginning; but it, nevertheless, allows for the important aerodynamic interactions which we wish to study. In particular, the blade twist, solidity, and taper ratio will enter the problem in important ways.

The next item on our analysis agenda is vibration and loads. These can enter the optimization through several paths. For example, there may be a minimum vibration requirement in which case it would enter as a constraint. On the other hand, additional vibration could require vibration attenuation devices which would add weight to the vehicle and thus degrade performance. In either case, vibratory

Table 1: Some Features of the Finite Element Program of Rotor Blade.

Elements:

1. Tapered and twisted beam elements are used.
2. Beam properties are specified at two ends of element and are varied linearly along element.
3. Lumped mass matrix with the effect of elastic offset included.
4. Stiffness terms include the following: bending, torsion, elongation, tension, kinetic energy due to inplane displacement, and "torsion-rotation" energy.

Capabilities of Program:

1. Rigid links are performed mathematically to increase efficiency of program and precision of results.
2. Discontinuity of beam properties is allowed.
3. Hinges of the blade are modeled "exactly".
4. Gradient information is calculated analytically.
5. Subspace iteration method is used to increase the efficiency of the iteration process.

airloads contribute to the fatigue of components and affect the life of blades. As with performance, we have elected to begin the optimization study with a simplified vibration analysis. Thus, in the initial stages, we are applying a given load spectrum to the blade with viscous damping added to simulate aerodynamic feedback. Obviously, this is a far cry from the detailed aeroelastic vibration analysis we plan to do later. However, it does give reasonable loads that interact with frequencies and mode shapes in a meaningful way. Therefore, it will allow a study of problems in combined optimization. It should also be noted that, since mode shapes enter the vibratory response, gradients of the modes are required in order to perform the optimization, see ref. 11.

The fifth item of analysis is that of stresses and fatigue life. Perhaps the most important goal of this present work is to incorporate this into the optimization process. In computation of stresses, we use the combined tensile stresses (due to centrifugal force) and the bending-torsion stresses due to the loading described above. These are combined in the conventional tensor way in order to find the three-dimensional "Mohr's Circle" that describes the stress state (static plus oscillatory). Fatigue life is then computed based on the methodologies described in refs. 13 and 14. In particular the infinite-fatigue-life stress is lowered by the typical "Endurance Reduction Factor" of 0.8 to obtain a "Reduced Endurance Limit." Next, a "Weighted Fatigue Approach" is used to modify the computed stresses. In this methodology, the alternating loads are multiplied by a factor of 2.0 and added to the static loads. A design is within the stress constraint if this computed maximum stress is less than the reduced endurance stress at all points within the structure. The use of stress as a constraint requires the computation of stress gradients. When primitive design variables are used in optimization, this method is straightforward, although involved. However, when the overall quantities are used, the detailed thicknesses are unknown at each iteration; and the stress computation encounters some necessary approximations. In this work, we also explore the consequences of these approximations.

The final element in the analysis is aeroelastic stability. Past work has shown this to be very important either when frequency placement is not a constraint or for hingeless and bearingless rotors. In this phase of our research, we are restricting ourselves to articulated and teetering rotors for which aeroelastic stability is not generally a problem. Thus, for the aeroelasticity portion of the analysis, we simply include a constraint that the center of mass be forward of the elastic axis of the blade at all cross sections. This prevents any classical bending-torsion flutter.

Objective Function

For the case of optimizing hover performance, it is possible to unify the entire problem into one objective function. This could be the payload (lifting force) of the rotor at a certain altitude at a given temperature. Such an objective function would be penalized by blade weight, by a reduction in rotor figure of merit, or by the added weight of vibration absorbers. To make this operational, one would have to decide on numerical values for the ratio of total mass to blade mass (taken above as 4 to 1) and on the number of pounds of isolator mass required to eliminate a certain number of pounds of 4/rev vibration at some location. Thus, the objective function would be the lifting capability at a given power minus the blade weight (multiplied by a factor to account for control-system weight to blade weight) minus the weight of vibration absorbers based on calculated vibrations. Similar objective functions could be formulated for other missions in terms of range or maneuverability.

RESULTS AND DISCUSSION

Scope of Present Results

This paper describes work in progress. However, we do have some very interesting numerical results obtained in the first phase of our work; and these results are presented here. In these first results, we have added the effect of stress constraints to the optimization, but we have not added the aerodynamic performance. Thus, the results below are for a fixed aerodynamic envelope. The initial blade design is the so-called "Hughes blade" of ref. 7 which resembles a typical McDonnell Douglas AH-64A blade in geometry but not in detailed design. The box-beam material is taken to be 6061 T6 Aluminum alloy with 35×10^3 psi yield stress and 13×10^3 psi endurance limit. Blade loads are considered harmonic in nature with a quadratic radial distribution. The strength of the zero harmonic is based on a given C_T , and other harmonics are given magnitudes based on the flight loads survey in ref. 15. The optimization is performed with CONMIN.

Section Properties

The first step in this optimization study was to try to match the physical data with the primitive cross-sectional variables. In the beginning work, we found that the optimizer indiscriminately placed stiffness in the box beam (rather than in the skin) which made skin thicknesses unreasonably small. Therefore, we fixed skin thickness and only allowed its modulus to vary. As a result, we could match all the physical stiffness properties with reasonable success, although the shear center seemed to end up closer to the front end of the box beam than in the data. Tables 2 and 3 outline the initial and modified procedures for this pre-optimization.

Table 2: Procedures Now Used to Determine Section Properties from Given Structural Data.

* Step 1: Design variables: t , s_1 , s_2 , or p (skin thickness)

$$\text{Objective: } w_1 (EA - EA_0)^2 + w_2 (EI_f - EI_{f0})^2 + w_3 (EI_c - EI_{c0})^2$$

$$\begin{aligned} \text{Constraints: } \quad & s_1 + s_2 < b, \quad GJ < GJ_0 \\ & (1-\tau_2)(EA_0) < EA < (1+\tau_1)(EA_0) \\ & (1-\tau_4)(EI_{f0}) < EI_f < (1+\tau_3)(EI_{f0}) \\ & (1-\tau_6)(EI_{c0}) < EI_c < (1+\tau_5)(EI_{c0}) \end{aligned}$$

* Step 2: Solve $G(\text{skin})$, so that $GJ = GJ_0$

* Step 3: Design variables: a , d , skin density.

$$\text{Objective: } w_1 (m - m_0)^2 + w_2 (\rho I_f - \rho I_{f0})^2 + w_3 (\rho I_c - \rho I_{c0})^2 + w_4 (e - e_0)^2$$

$$\begin{aligned} \text{Constraints: } \quad & (1-\tau_2)(m_0) < m < (1+\tau_1)(m_0) \\ & (1-\tau_4)(\rho I_{f0}) < \rho I_f < (1+\tau_3)(\rho I_{f0}) \\ & (1-\tau_6)(\rho I_{c0}) < \rho I_c < (1+\tau_5)(\rho I_{c0}) \\ & (1-\tau_8)(e_0) < e < (1+\tau_7)(e_0) \end{aligned}$$

* Where w_i is weighting scalar, τ_i is tolerant value, $()_0$ is given data, $()_f$ represents quantity in flapping direction, $()_c$ represents quantity in chordwise direction, and e is elastic offset.

Table 3: Proposed Procedures to Determine Section Properties from Given Structural Data.

* Step 1: Design variables: t, s_1, s_2 (Given skin thickness & $G(\text{skin})$)

$$\text{Objective: } w_1 (EA - EA_0)^2 + w_2 (EI_f - EI_{f0})^2 + w_3 (EI_c - EI_{c0})^2 + w_4 (GJ - GJ_0)$$

$$\begin{aligned} \text{Constraints: } & s_1 + s_2 < b \\ & (1-\tau_2)(EA_0) < EA < (1+\tau_1)(EA_0) \\ & (1-\tau_4)(EI_{f0}) < EI_f < (1+\tau_3)(EI_{f0}) \\ & (1-\tau_6)(EI_{c0}) < EI_c < (1+\tau_5)(EI_{c0}) \\ & (1-\tau_8)(GJ_0) < GJ < (1+\tau_7)(GJ_0) \end{aligned}$$

* Step 2: If the results from Step 1 are not acceptable, then the another skin thickness and $G(\text{skin})$ are selected and Step 1 is performed again. Repeated Steps 1 & 2 until reasonable data are obtained.

* Step 3: Design variables: a, d (Given skin density)

$$\text{Objective: } w_1 [m - m_0]^2 + w_2 [(\rho I_f + \rho I_c) - (\rho I_{f0} + \rho I_{c0})]^2 + w_3 [e - e_0]^2$$

$$\begin{aligned} \text{Constraints: } & (1-\tau_2)(m_0) < m < (1+\tau_1)(m_0) \\ & (1-\tau_4)(\rho I_{f0}) < \rho I_f < (1+\tau_3)(\rho I_{f0}) \\ & (1-\tau_6)(\rho I_{c0}) < \rho I_c < (1+\tau_5)(\rho I_{c0}) \\ & (1-\tau_8)(e_0) < e < (1+\tau_7)(e_0) \end{aligned}$$

Next we optimized the blade for frequency placement using both the overall properties (EI , GJ , etc.) and the primitive variables (t , s , etc.). Here, we found that we could optimize the blade by either method and then restore overall properties to primitive values by the properties optimization methodology discussed above. Table 4 summarizes the three phases of this process. In the first two phases, various frequencies are brought within the desired windows by the use of frequency placement in the objective function. Then, with all frequencies within these windows, the windows become constraints and weight is minimized. Table 5 shows the result of this optimization when the structural properties (primitive variables) are used. A total of 77 iterations are required to meet all requirements. Table 6 shows the identical optimization when the physical properties (EI , etc.) are used. In this case, an optimum is reached in only 61 iterations. Thus, there is some saving in not using primitive variables. However, one has the added problem of turning these overall quantities into primitive variables in order to realize the design. Furthermore, one cannot apply stress constraints at each iteration if the internal geometry is not known. Therefore, in the work to follow, we concentrate on optimization with primitive variables.

Optimization with Stress Constraints

Next, we added stress constraints to the optimization process. In the beginning of this phase, when we only considered yield stresses, we found that the stress constraint never became active. In other words, the optimization to place frequencies never did anything so drastic to the blade that any section would reach yield stress. However, when we extended the stress constraints to include fatigue life, stresses became an important part of the analysis.

Table 4: Procedures of Optimizing the Hughes Articulated Rotor Blade.

Based on:	<u>Structural Properties</u>	<u>Physical Properties</u>
<u>Frequency placement (I)</u>		
Design variables:	$EI_f, EI_c, m, \rho I_f, \rho I_c$	t, s_1, s_2, a, d
Constraints	$1.0 < p(\text{flapping-1st})^* < 1.5$ $2.3 < p(\text{flapping-2nd}) < 2.75$ $0.3 < p(\text{inplane-1st}) < 0.7$ $4.23 < p(\text{torsion-1st}) < 4.7$ $1.2581 \times 10^4 \text{ (mugs-in}^2\text{)} < \text{autorotation}$ side constraints on design variables	
Objective	$[p(\text{flapping-3rd}) - 4.5]^2 + [p(\text{inplane-2nd}) - 6.5]^2 + [p(\text{flapping-4th}) - 7.5]^2$	
<p>* $p = (\text{blade natural frequency}) / (\text{rotor rotational frequency})$.</p>		

Based on:	<u>Structural Properties</u>	<u>Physical Properties</u>
<u>Frequency placement (II)</u>		
Design variables:	$EI_f, EI_c, GJ, m, \rho I_f, \rho I_c$	t, s_1, s_2, a, d
Constraints	$1.0 < p(\text{flapping-1st}) < 1.5$ $2.3 < p(\text{flapping-2nd}) < 2.7$ $4.3 < p(\text{flapping-3rd}) < 4.7$ $7.3 < p(\text{flapping-4th}) < 7.7$ $0.3 < p(\text{inplane-1st}) < 0.7$ $6.3 < p(\text{inplane-2nd}) < 6.7$ $4.25 < p(\text{torsion-1st}) < 4.7$ $4.3 < p(\text{torsion-1st}) < 4.7$ $12.3 < p(\text{torsion-2nd}) < 12.7$ $1.2581 \times 10^4 \text{ (mugs-in}^2\text{)} < \text{autorotation}$ side constraints on design variables	
Objective	$([p(\text{flapping-5th}) - 11.5]^2 + [p(\text{torsion-2nd}) - 12.5]^2)$	

Table 4: Procedures of Optimizing the Hughes Articulated Rotor Blade (Concluded).

Based on	<u>Structural Properties</u>	<u>Physical Properties</u>
<u>Weight Minimization</u>		
Design variables:	$EI_f, EI_c, GJ, m, \rho I_f, \rho I_c$	t, s_1, s_2, a, d
Constraints	$1.0 < p(\text{flapping-1st}) < 1.5$ $2.3 < p(\text{flapping-2nd}) < 2.7$ $4.3 < p(\text{flapping-3rd}) < 4.7$ $7.3 < p(\text{flapping-4th}) < 7.7$ $11.3 < p(\text{flapping-5th}) < 11.7$ $0.3 < p(\text{inplane-1st}) < 0.7$ $6.3 < p(\text{inplane-2nd}) < 6.7$ $4.3 < p(\text{torsion-1st}) < 4.7$ $12.3 < p(\text{torsion-2nd}) < 12.7$ $0. < \text{elastic offset}^*$ $1.2581 \times 10^4 (\text{mugs-in}^2) < \text{autorotation}$ side constraints on design variables	
Objective	Weight of Blade	

* Except at station 75 which originally has an elastic offset equal to -.52 and has the constraint (elastic offset > -.54).

Table 5: Optimization Results of Hughes Blade (Structural Properties as Design Variables).

	Original	Frequency Placement(I)	Frequency Placement(II)	Weight Minimization
Weight (lbs)	203.42	217.36	217.03	193.03
Autorotation	12581.	13233.	13203.	12661.
p(inplane-1st)	0.4756	0.4820	0.4820	0.4677
p(flapping-1st)	1.0293	1.0304	1.0304	1.0257
p(flapping-2nd)	2.7451	2.6143	2.6287	2.6876
p(torsion-1st)	4.2483	4.2501	4.2563	4.3058
p(flapping-3rd)	4.9035	<u>4.5101</u>	4.5772	4.6979
p(inplane-2nd)	6.8914	<u>6.5029</u>	6.5118	6.6958
p(flapping-4th)	7.9378	<u>7.5026</u>	7.5923	7.5558
p(flapping-5th)	12.058	10.933	<u>11.500</u>	11.302
p(torsion-2nd)	12.921	12.927	<u>12.500</u>	12.642
p(flapping-6th)	16.996	15.521	15.921	15.484
Iterations(CONMIN)	--	21	7	49

Table 6: Optimization Results of Hughes Blade (Physical Properties as Design Variables).

	Original	Frequency Placement(I)	Frequency Placement(II)	Weight Minimization
Weight (lbs)	203.42	213.86	214.55	203.15
Autorotation	12581.	12579.	12777.	12656.
p(inplane-1st)	0.4756	0.4841	0.4824	0.4708
p(flapping-1st)	1.0293	1.0310	1.0305	1.0269
p(flapping-2nd)	2.7451	2.5801	2.6049	2.6989
p(torsion-1st)	4.2483	4.3448	4.3345	4.3014
p(flapping-3rd)	4.9035	<u>4.5049</u>	4.5779	4.6985
p(inplane-2nd)	6.8914	<u>6.5020</u>	6.5186	6.6272
p(flapping-4th)	7.9378	<u>7.4962</u>	7.6803	7.6934
p(flapping-5th)	12.058	11.151	<u>11.500</u>	11.553
p(torsion-2nd)	12.921	12.580	12.684	12.380
p(flapping-6th)	16.996	15.850	16.243	16.293
Iterations(CONMIN)	--	16	7	38

Figure 3 shows the critical stresses along the blade for the original design (i.e., the schematic model of the blade with one cell in the spar). The dashed line is the static stress in hover which comes primarily from tension stress and bending moments. Notice that the moment must go to zero at the hinge and at the tip, but the tension is zero only at the tip. The solid line is the dynamic stress from our oscillatory vertical loading distribution. This loading, based on ref. 15, includes up to 8 harmonics; and the oscillatory part is doubled as per the fatigue methodology in refs. 13 and 14. The solid curve with open symbols is the reduced fatigue-life stress discussed earlier. We can see that the original blade meets the fatigue-life criterion except near the hinge.

Figure 4 shows the stress distribution on the optimized blade for which frequencies have been placed in predetermined windows but without stress constraints, as in ref. 7. One can see that the slight overstress near the hinge ($r=50$) still exists. However, a large overstress has developed at $r=240$. This is due to a lower thickness which was placed there to lower the frequencies of the 3rd flapping and 2nd torsional modes. Therefore, this previously obtained optimum has reduced fatigue life. Starting with this solution, we began a second optimization with the fatigue criteria as side constraints. Figure 5 shows the results for the newly optimized blade. We can see that the optimizer is able to maintain the frequencies within the desired windows and still satisfy the fatigue life constraints. The constraint is active near the root and at the soft section. Furthermore, although extra structural material has been added to lower stresses, this allows added mass to be removed (the autorotational constraint) so that the final design is no heavier than the one that violated the stress constraints.

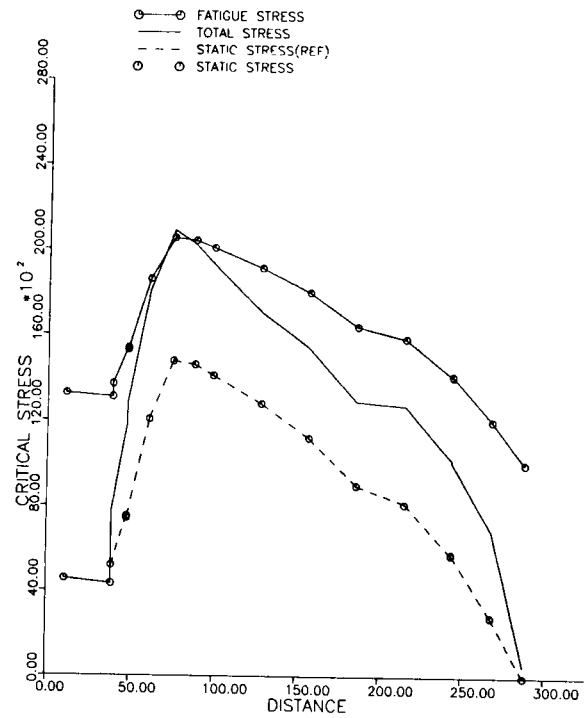


Figure 3. Stresses of Initial Design

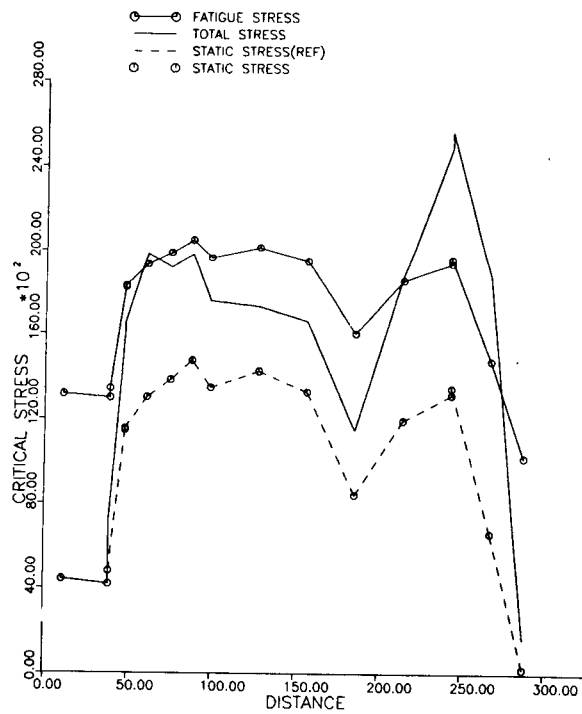


Figure 4. Stresses of Optimum Design

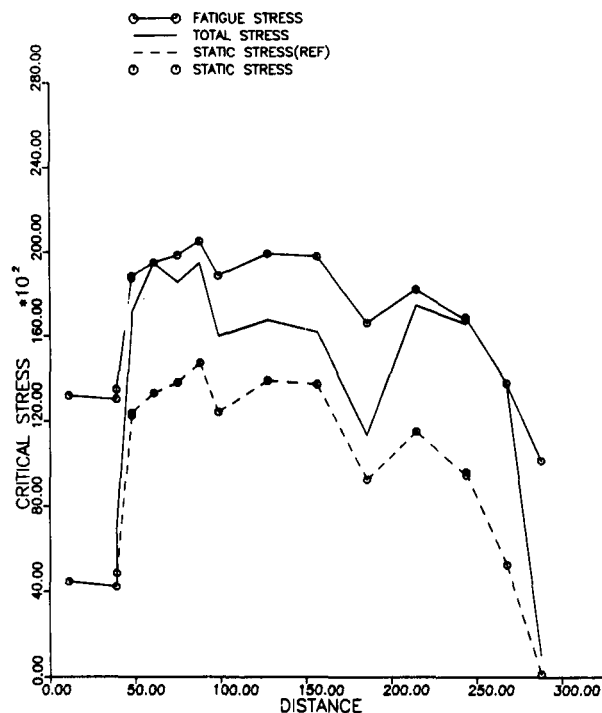


Figure 5. Stresses after Stress Constraint

Figures 6-10 show the major primitive variables (t , s_1 , s_2 , a , and d) before optimization (diamonds), after frequency placement (dashed line), and after application of stress constraints (solid line). Looking first at thickness, fig. 6, we see that (after frequency placement) the thickness has been drastically reduced near station 240. However, once the stress constraint is applied, this thickness is returned to its original value at the point of high stress (station 240) but not further inboard where modal curvature is highest. Therefore, there is no need to compensate for this added stiffness (which occurs primarily in GJ and EI_{yy}). One also notices that a large increase in thickness occurs at to the tip. As pointed out in ref. 7, near the tip there is no real distinction between structural mass and lumped mass because structure is ineffective. Consequently, the lumped mass added to the tip (to minimize weight for a given inertia) has been placed in box-beam mass rather than in non-structural mass, figs. 9 and 10. The decrease in web thickness with frequency placement, seen in figs. 7 and 8, does not impact the fatigue life because it occurs away from the high-stress areas. Thus, little change occurs in s_1 or s_2 after the stress constraint is applied.

It should be noted here that the addition of the stress constraints increases the computational time required to optimize by a factor of 5 to 6. The increased time is not so much in the stress computation (which is very simplified here). Rather, the computational time is expended in moving from unfeasible to feasible solutions and in calculating the complicated modal sensitivities that are needed for gradients of the stress constraints. Thus, more research must be done in these areas before more sophisticated stress and aeroelastic constraints can be applied.

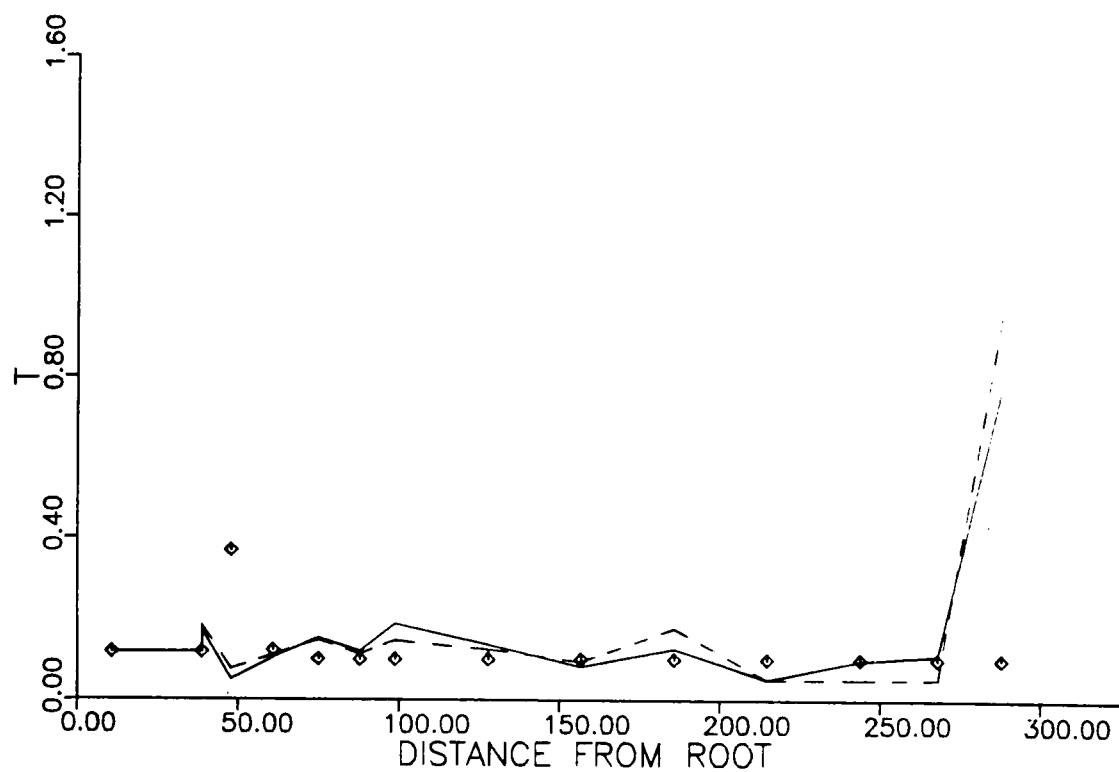


Figure 6. Flange Thickness, t

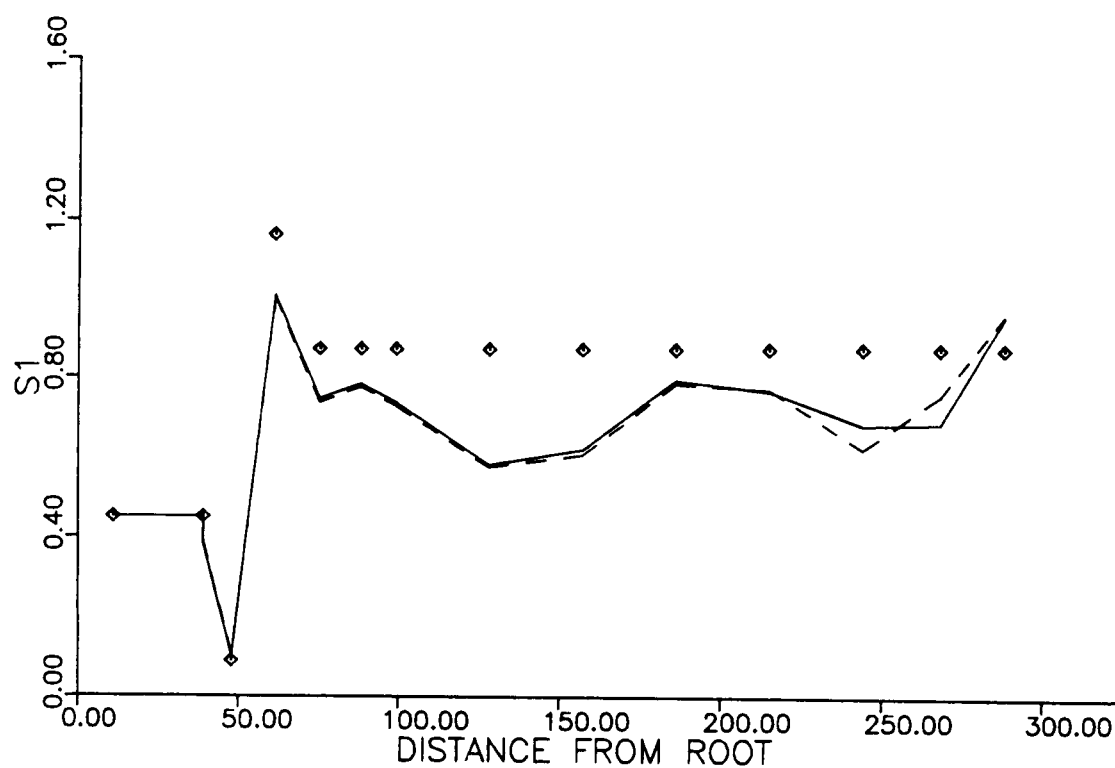


Figure 7. Webb Thickness, s_1

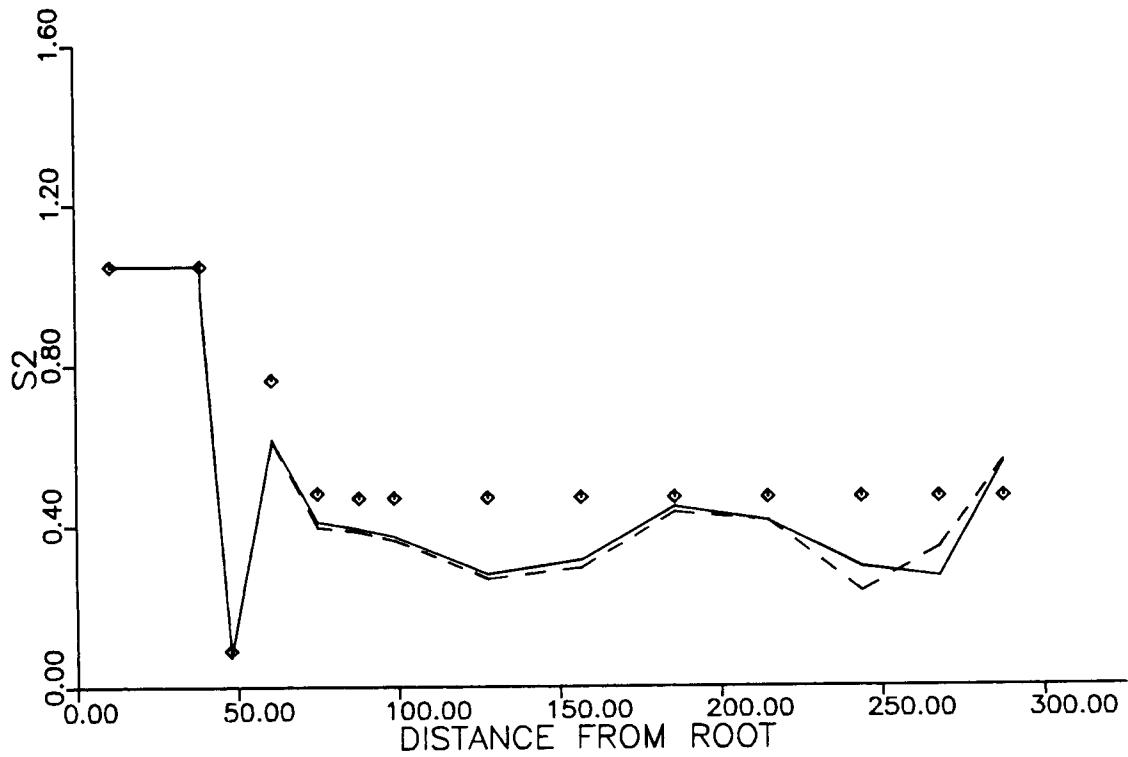


Figure 8. Webb Thickness, s_2

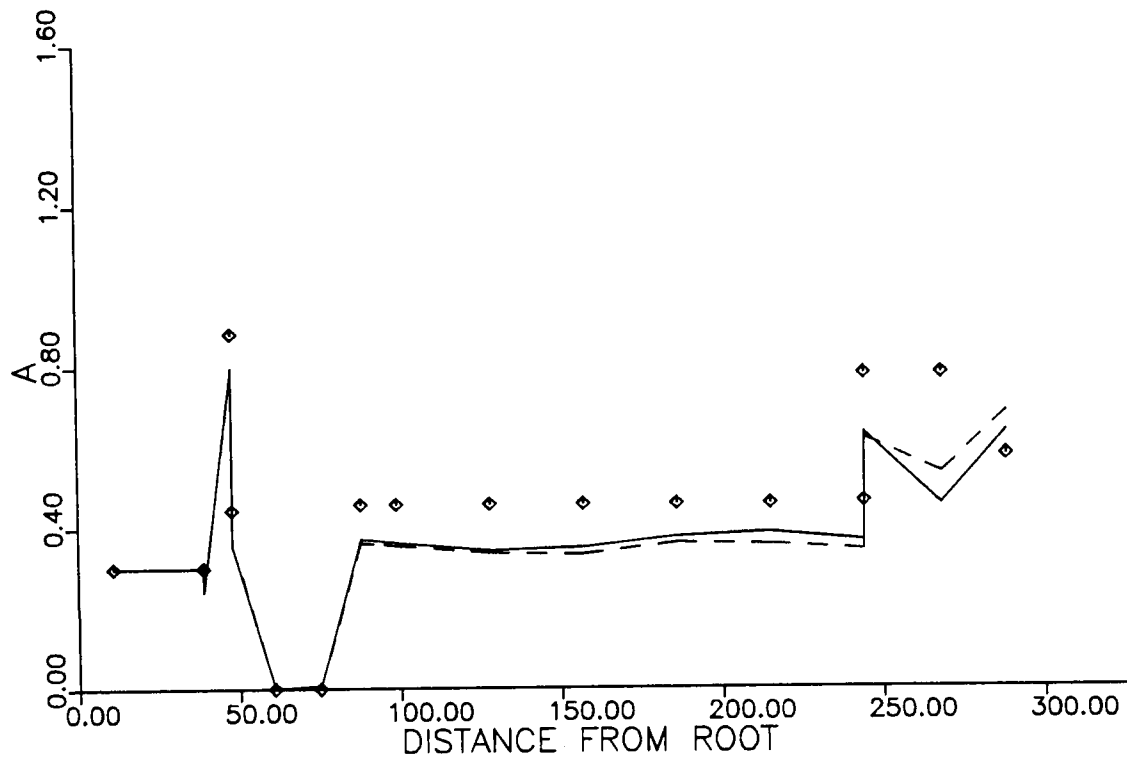


Figure 9. Tip Mass Size, a

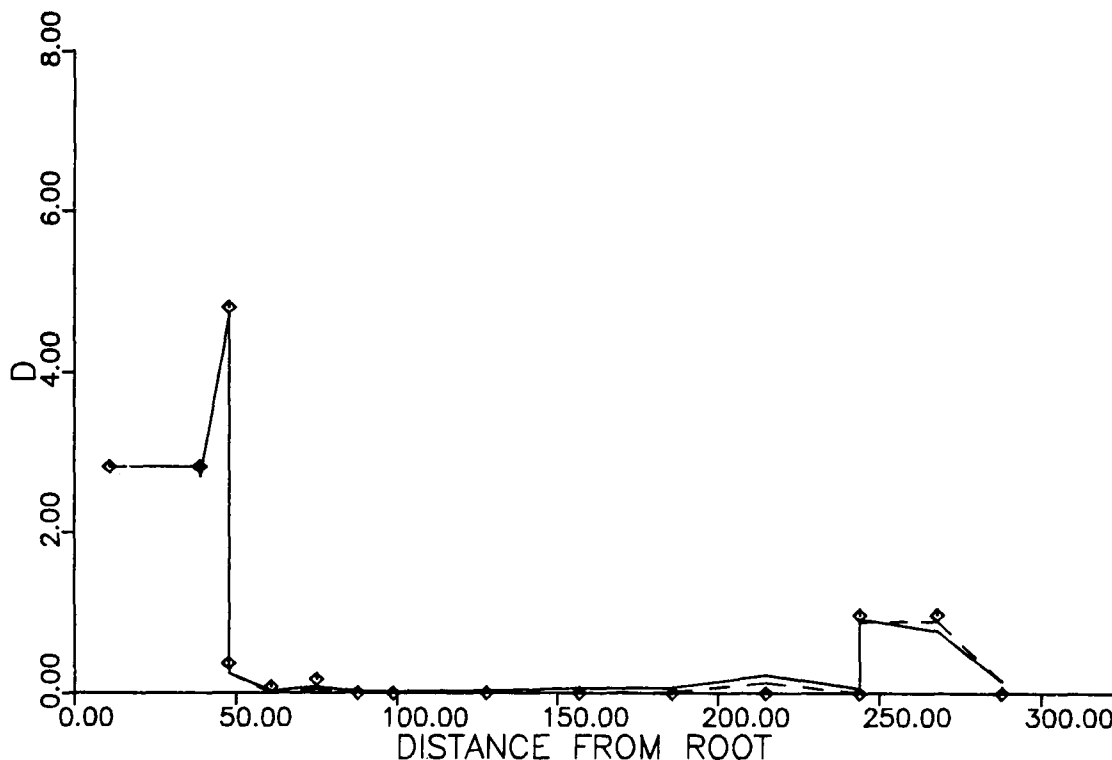


Figure 10. Internal Mass Size, d

SUMMARY AND CONCLUSIONS

In this paper, we have outlined general principles whereby multidisciplinary optimization could be performed in the design of helicopter blades. A very important part of the problem is the development of efficient computational schemes for the various components of the analysis. Results show that stress constraints based on fatigue life can be added to conventional structural optimization. However, we are still a long way from a completely integrated, automated design process.

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